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TRIBOLOGICAL EVALUATION OF CONTACTS LUBRICATED BY OIL-REFRIGERANT MIXTURES

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ABSTRACT

The tribological characteristics of the most common contact geometries found in compressors of air conditioning and refrigeration systems have been experimentally investigated by means of a unique high pressure tribometer (HPT). The HPT has been used to experimentally simulate the friction and wear behavior of various metal contact pairs lubricated by oil-refrigerant mixtures in environments found in compressors. The refrigerants used in this program are CFC-12 to obtain baseline data and its prime replacement candidate, HFC-134a. The CFC-12 has been tested with mineral oils and synthetic alkylbenzenes while the HFC-134a has been tested with monoether polyalkylene glycol (PAG's) and pentaerythritol polyolester oils. Since the amount of refrigerant dissolved in the oil is a function of both pressure and temperature, and the friction and wear of a given contact can be significantly affected by the concentration of refrigerant in the oil, the friction and wear data obtained from this test program should be a good indicator of what can be expected in compressors.

INTRODUCTION

Data on viscosity, miscibility, and other properties for commonly used oil-refrigerant mixtures are given by Little (1), Parmelle (2), Spauschus (3), and Spauschus and Speaker (4); while general lubrication requirements are given in Grim (5), and Spauschus (6). However, the integrity of a tribo-contact in a compressor is mainly influenced by the tribological properties of the oil-refrigerant mixture. Although it is widely known that refrigerant vapor under pressure will tend to saturate into the oil, the data on the resulting tribological properties of the mixture are less complete. Much of the tribological data in the literature have been obtained by using standard specimen screening tests where the contact is submerged in oil through which bubbling refrigerant is fed during testing. Although this approach will result is some refrigerant saturating in the oil, the environment existing in a compressor is not simulated. To better simulate environments in compressors, Komatsuzaki et al (7) and Komatsuzaki & Homma (8) have evaluated oil-refrigerant mixtures in a HPT. However, the pressure capabilities of their machine is not capable of modeling pressures found in some compressors. Since pressure directly effects the amount of refrigerant saturated into the oil and the tribological properties of the resulting mixture are highly dependent on the amount of refrigerant in the oil, it is critical that the environment existing in compressors be duplicated in any specimen testing program. A HPT, developed over the past two years as part of the Air Conditioning and Refrigeration Center at the University of Illinois, is capable of simulating environments (temperature and pressure) found in almost all compressors. The main purpose of this paper is to describe this HPT and to present friction and wear data for some material contact pairs and oil-refrigerant mixtures of current interest. It is hoped that the specimen data obtained with this HPT more accurately correlate with actual components behavior in compressors.

HIGH PRESSURE TRIBOMETER

The HPT used in this investigation is schematically shown in Fig. 1. In order to allow for a pressurized refrigerant environment, the test must occur within the confines of a pressure chamber. The chamber on the HPT consists of two halves that are separated to permit cleaning, specimens mounting and supplying oil to an oil cup. The upper half of the chamber is stationary, while the lower half can be raised and lowered by the Z-axis servo motor. When closed, the two halves are sealed together by means of a six-inch custom made telescopic seal. This seal is capable of withstanding a test pressure of up to 1.725 MPa (250 psig), as well as vacuums of less than 100 microns. The only other seal required in the chamber is a custom made dynamic rotary seal for the spindle. This seal provides dry sealing in a refrigerant environment for pressures of up to 1.725 MPa (250 psig) and speeds of up to 2000 rpm. There is also a v-ring seal used in conjunction with the rotary seal to maintain vacuum integrity. This chamber completely encloses the specimens and test apparatus during testing.

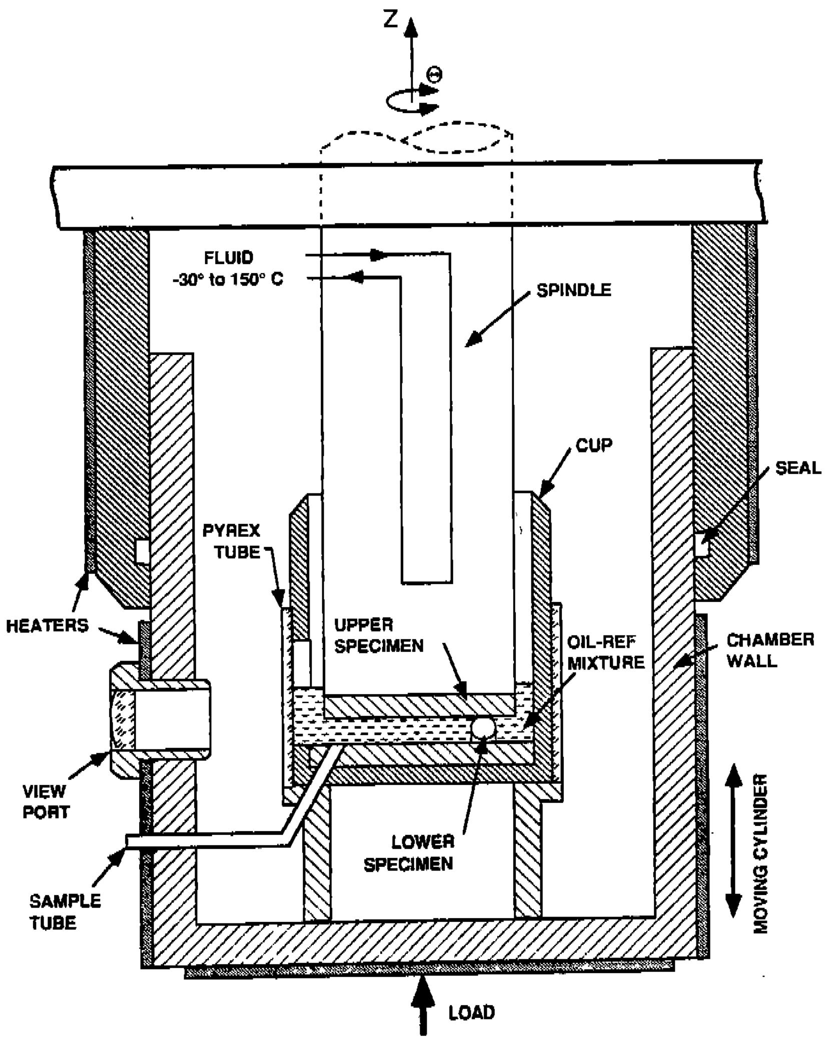

Fig. 1 - Tribometer Section

One of the more beneficial aspects of this test machine is the wide variety of contact geometries that can be tested. Whereas most friction and wear machines are designed to test one specific contact, the HPT, with the appropriate specimen holders, can accept virtually any type of contact pair which needs to be tested under simple sliding conditions. The upper specimen is secured to the spindle and thus models the moving part of the contact. The cup, which is secured in the lower half of the chamber, holds the lower stationary specimen. Three sight ports machined through the wall of the cup, corresponding to sight ports in the chamber walls, allow for viewing of the contact during testing. A Pyrex glass sleeve, sealed at the bottom by an o-ring, surrounds the cup. The cup can be filled with a lubricant, completely submerging the contact to be tested.

Thermally, the tribometer has several important features. Virtually all internal surfaces of the chamber can be heated. This is required to prevent condensation of refrigerant on these surfaces at the high test pressures. The heaters consist of one 400 W cartridge in the top half of the chamber, and two 500 W cartridges in the lower half. The temperature of the upper specimen holder is controlled by an external recirculating unit which is capable of maintaining temperatures from -30° C to 150° C (-20° F to 300° F). The temperature is controlled by pumping a heat transfer fluid through the spindle. The high value of the heat transfer coefficient of the fluid and the unique design of the passages in the spindle act to maintain a constant test temperature. A separate chiller maintains critical parts of the tribometer at ambient temperature. All of these heaters have independent controls and the temperatures can be monitored on the main control panel.

The loading and spindle motion of the HPT are controlled by two servo motors. The motion of the upper specimen (spindle) is generated by a large Θ-axis servo motor. This motor is capable of simple unidirectional rotation (0-2000 rpm) and oscillatory motion (up to 5 Hz). Although the on-board controls of the HPT only allow for sinusoidal or triangular waveforms, the system is currently being updated to permit any waveform. The position of the Θ-axis is monitored by a differential optical encoder with a resolution of 0.1°. The Z-axis motion is provided by a lead screw which is driven by the Z-axis servo motor via a backlash-free 100:1 harmonic drive. This motor controls the motion of the lower half of the chamber as well as providing axial loads, up to 4450 N (1000 lb). A unique internal diaphragm spring suspension system allows the motor to accurately apply low loads while the chamber is under high pressure.

Feedback for the axial load is provided through a complex transducer outfitted with strain gages. The transducer provides feedback in the form of F_x, F_y, F_z, and M_z. Each of the force (torque) directions has its own independent amplifier that excites the strain gages. This allows for each direction to be set to the appropriate sensitivity so that the accuracy of the force reading can be improved. When the chamber is pressurized to 1725 MPa (250 psig), it takes approximately 31 kN (7000 lbs) to hold the two halves closed. Most of this force is taken up in the suspension system so that, with proper amplifier configuration, test loads as low as one pound can accurately be applied and monitored.

The HPT has also been outfitted with apparatuses for purging, charging and sampling as shown in Fig. 2. Two vacuum pumps work in tandem to purge the system. The larger of the two pumps is used to purge the main chamber and external lines. The other pump is to remove any vapor outgassed from the grease in the main bearings. An external 8 pound pressure vessel is used to charge the chamber with refrigerant. A silicone heating blanket around the vessel is used to transfer the pressurized refrigerant to the chamber. A 30 lb refrigerant tank, attached to the pressure vessel by a quick disconnect, is used to supply refrigerant to the vessel. The chamber is also outfitted with a 50 cc sample cylinder which can be used to siphon the oil-refrigerant mixture sample during a test. This sample is used to determine the approximate amount of refrigerant saturated in the oil as well as possible oil degradation. A separate 15 lb refrigerant tank serves as a drain tank which collects used refrigerant so that it can be recycled.

Test data were collected through the use of a COM based (RS-232c) data acquisition system. A personal computer communicates with the motherboard of the HPT using SECS-I communication protocol. The PC was used to configure the strain gage amplifiers as well as to read the loads, position, speed, and temperature data during the test. The data are read directly to a file on disk so that later they can be imported as a numeric file in Lotus 1-2-3™.

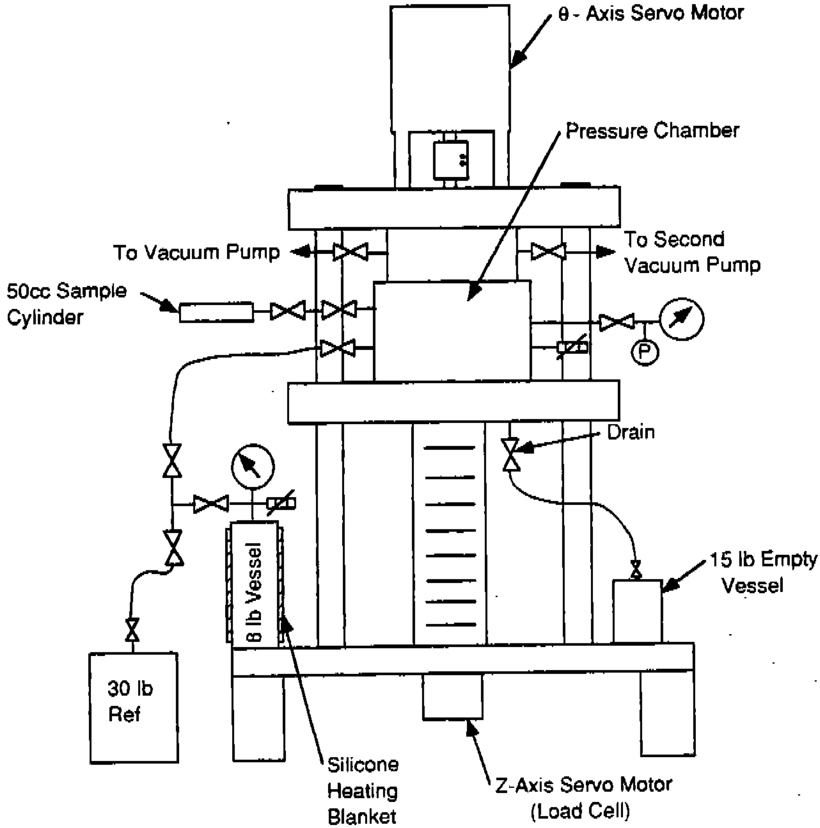

Fig. 2 - Tribometer Schematic

EXPERIMENTAL METHOD

Contacts Under Study

Table 1 shows the types of contact geometries and material contact pairs evaluated. The counterformal, area, and conformal contact geometries are representative of those found in actual compressors. Most of the operating and environmental conditions are also typical of those that are currently used in compressors. The critical contacts under study are those simulating the vane-piston contact in the rolling piston, the shoe-plate contact in the swash plate, and the wrist pin-bearing contact in the reciprocating piston. These three geometries represent counterformal, area, and conformal contacts respectively. The vane piston contact is simulated by a stationary hardened tool steel pin rubbing against an oscillating hardened cast iron plate. The shoe-plate contact is simulated by a stationary circular bronze shoe rubbing against a rotating hardened ductile cast iron plate. Finally, the wrist pin-bearing contact is simulated by a stationary case-hardened mild steel pin rubbing against an oscillating aluminum pad. The specific motions and operating conditions for these contacts are shown in Table 2. With the exception of the swash plate, each test ran at load, speed, pressure, and temperature representative of those found in actual compressors. To avoid hydrodynamic lift-off and generate measurable wear, it was necessary to run the area contact at higher loads and slower speeds than what is typically encountered in the swash plate compressor.

Table 1 - Contact Geometries & Materials

Description	Counter-formal Contact	Area Contact	Conformal Contact
Geometry • **Upper**	76.2 mm Ø Flat Disk	76.2 mm Ø Flat Disk	76.2 mm Ø Flat Disk
• **Lower**	6.35 mm Ø Pin L=9.53 mm	5.08 mm Ø Flat Shoe	6.35 mm Ø Pin with 1220 mm Ø L=9.53 mm
Materials • **Upper**	Gray C.I.	Ductile C.I.	Die Cast Al
• **Lower**	Tool Steel	Bronze	Mild Steel
Hardness • **Upper**	50 Rc	42 Rc	---
• **Lower**	65 Rc	---	63 Rc
Surface Topography • **Upper**	Ground	Ground	Ground
• **Lower**	Ground	Lapped	Ground
Surface Finish (Ra) • **Upper**	0.13 µm	0.13 µm	0.26 µm
• **Lower**	0.13 µm	0.21 µm	0.10 µm

Table 2 - Operating Conditions

Operating Conditions	Counter-formal Contact	Area Contact	Conformal Contact
Contact Load (MPa)	1,034	124 (P/A)	13.8 (P/LD)
Type of Motion	Oscillatory	Unidirect.	Oscillatory
Speed (m/sec)	± 0.51 max	0.20	±0.17 max
Angular Amplitude	± 50°		± 20°
Angular Frequency	5 Hz		4 Hz
Env. Pressure (MPa)	1.55	0.172	0.172
Env. Temp (°C)	80.6°	73.9°	100°
Test Duration	1 hr	1 hr	1 hr

Lubricants Under Study

The lubricants that have been evaluated are classified into four types: mineral oils, alkylbenzenes, polyalkylene glycols (monoether), and polyolesters (pentaerythritol). The first two are used with CFC-12 for obtaining baseline friction and wear data, while the latter two are the more promising lubricants for use with HFC-134a. The mineral oils tested are presently used in the swash plate and reciprocating piston compressors, while the synthetic alkylbenzene is used in the rolling piston compressor. Where possible, both base and formulated versions of each lubricant were tested. Although proprietary in nature, the formulated oils are versions of the respective base oil with an additive package to improve lubricative properties. Some of the relevant lubricant properties are shown in Table 3. The mineral, alkylbenzene, and polyolester oils were fully miscible with their respective refrigerant, while the PAG oil was only partially miscible.

Table 3 - Lubricant Data

Oil Number	Oil Type	Family [1]	Additives	Viscosity μ (cS) @40° C	Viscosity μ (cS) @100° C
Min1	Mineral Oil	-	No	102	11.12
Min2	Mineral Oil	-	No	12	2.6
Alkbenz-B	Alkylbenzene	-	No	57	5.8
Alkbenz-F	Alkylbenzene	-	Yes	57	5.8
PAG1-B	Polyalkylene glycol	Mono	No	135	25
PAG1-F	Polyalkylene glycol	Mono	Yes	135	25
PAG2-B	Polyalkylene glycol	Mono	No	100	20
PAG2-F	Polyalkylene glycol	Mono	Yes	100	20
Est1-B	Polyolester	PE	No	23.94	4.88
Est1-F	Polyolester	PE	Yes	23.9	4.87
Est2-B	Polyolester	PE	No	91.37	10.19
Est2-F	Polyolester	PE	Yes	91.4	10.18
Est3-B	Polyolester	PE	No	11.5	2.8
Est3-F	Polyolester	PE	Yes	11.5	2.8

[1] PE- Pentaerythritol ester
Mono- Monoether

Table 4 - Amount of Refrigerant Saturated in Lubricant

Oil Number	Ref Type	Miscibility	Contact Type	Temp (°C)	Press. (MPa)	Weight % Ref in Oil
Min1	R12	Full	Area	73.9	0.172	4.9
Min2	R12	Full	Conformal	100	0.172	1.7
Alkbenz-B	R12	Full	Counterformal	80.6	1.550	42.5
Alkbenz-F	R12	Full	Counterformal	80.6	1.550	41.1
PAG1-B	R134a	Partial	Area	73.9	0.172	3.1
PAG1-F	R134a	Partial	Area	73.9	0.172	2.5
PAG2-B	R134a	Partial	Conformal	100	0.172	1.8
PAG2-F	R134a	Partial	Conformal	100	0.172	---
PAG2-B	R134a	Partial	Counterformal	80.6	1.550	17.1
PAG2-F	R134a	Partial	Counterformal	80.6	1.550	21.2
Est1-B	R134a	Full	Counterformal	80.6	1.550	32.9
Est1-F	R134a	Full	Counterformal	80.6	1.550	28.5
Est2-B	R134a	Full	Area	73.9	0.172	0.3
Est2-F	R134a	Full	Area	73.9	0.172	0.5
Est2-B	R134a	Full	Conformal	100	0.172	1.4
Est2-F	R134a	Full	Conformal	100	0.172	1.9
Est3-B	R134a	Full	Conformal	100	0.172	1.3
Est3-F	R134a	Full	Conformal	100	0.172	0.9

PROCEDURES

In order to assure repeatability of results, the same testing procedure was used for all tests. Each test was repeated at least once to compute the average friction and wear. Before each test, the cup and specimens were ultrasonically cleaned with suitable solvents and then rinsed with 2-propanol to remove any remaining residues. After the specimens and oil were installed, the chamber was purged to at least 300 microns. For tests involving oils, the chamber could only be purged down to about 300 microns due to vaporization of the lubricant; while for tests with refrigerant alone, the chamber was purged to better than 100 microns. The chamber was charged with refrigerant by an 8 lb pressure vessel as shown in Fig. 2. The temperature of the vessel was raised to generate sufficient internal pressure before opening the valves to transfer vapor refrigerant from the pressure vessel to the HPT. To permit the refrigerant to fully saturate into the oil, the refrigerant-oil mixture in the HPT was maintained at thermal and pressure equilibrium for one hour prior to initiating the test . Table 4 also shows approximate amounts of refrigerant saturated in the oil. These data were obtained by sampling the oil-refrigerant mixture and slowly letting the refrigerant evaporate (based on ASHRAE standard (9)). All tests ran for 60 minutes except where seizure occurred.

Wear results are based on measurements taken immediately after completion of each test except for the area contact. The bronze shoe from the area contact was ultrasonically cleaned and allowed to dry in a desiccant chamber to remove any moisture prior to weighing. The amount of wear shown in Fig. 3 is the difference between the weight of the shoe prior to testing and the weight afterwards. The amount of wear for the conformal contact was obtained by measuring the wear scar depth in the aluminum pad with a Talysurf 10 surface profiler. The counterformal contact wear was obtained by measuring the wear scar width on the surface of the tool steel pin with an optical microscope. X-ray Photoelectron Spectroscopy (XPS) was used to determine the existence, if any, of surface films formed during testing.

RESULTS

Counterformal Contact

Wear data for the counterformal contact are shown in Fig. 3. The formulated versions of the three lubricants (Alkbenz-F, Est1-F, and PAG2-F), by themselves, provide better wear resistance than their base counterparts. When R12 was added to the base alkylbenzene, wear decreased. XPS analysis of the tool steel pin showed that iron chloride ($FeCl_2$) surface films were formed. The hypothesis is that these films essentially act as EP agents in the oil thus reducing wear. The formulated alkylbenzene does not show a comparable improvement when tested with R12, probably due to the additives which already exist in the oil.

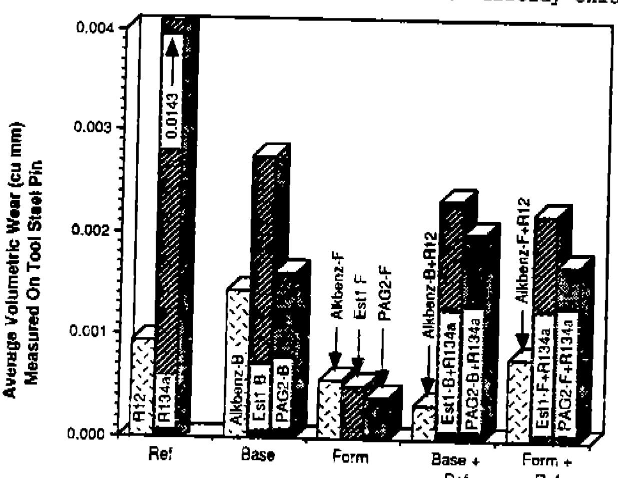

Fig. 3 - Counterformal Contact Wear Results

While R12, by itself, provides very good wear resistance, testing in an R134a environment shows extremely high wear rates, similar to testing in air. XPS analysis of the pins tested in R134a environments showed that no surface films were produced. Also, it is interesting to note that the formulated ester and PAG do not show improvements in wear resistance over their respective base when R134a is added. Testing of both the esters and PAGs with R134a tends to increase wear. This is due to the fact that the addition of R134a to the lubricant decreases its effective viscosity. This lower viscosity decreases the probability of generating protective oil films between surfaces, thus producing higher wear.

The coefficient of friction for the counterformal contact is shown in Fig. 4. As with the wear, the coefficient of friction is highest for R134a by itself. Overall, only slight variations in coefficient of friction are observed for all tests conducted The tests involving lubricants with R134a show slightly higher values of the coefficient of friction than the corresponding tests without R134a.

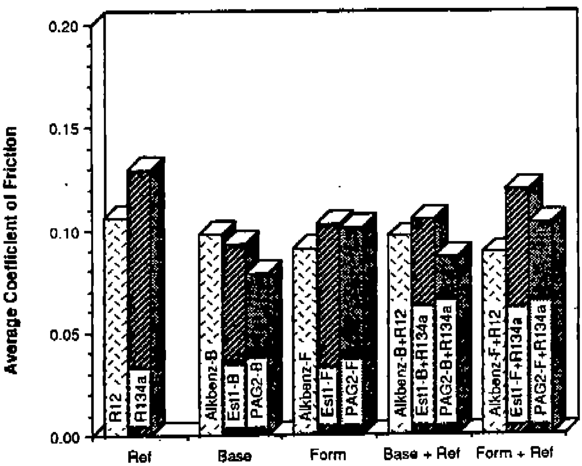

Fig. 4 - Counterformal Contact Coefficient of Friction

Area Contact

The wear associated with the area contact is shown graphically in Fig. 5. When the base oils are used alone, the mineral oil seems to provide the best wear resistance. Although the two formulated oils by themselves give better wear characteristics than the base mineral oil, once refrigerants are added to the lubricants, the mineral oil-R12 mixture provides the best wear resistance. The presence of R12 promotes chemical reaction on the bronze shoe, producing copper chloride ($CuCl_2$) and very small amounts of zinc fluoride (ZnF_2) as surface films. As with the counterformal contact, the surface films help to protect the surface and therefore lower wear. Although not verified, the R12 most likely formed $FeCl_2$ surface films on the mating ductile cast iron disk as well (7). The wear resistance of base PAG and ester oils in a R134a environment is much lower than that for mineral oil in R12. From the limited number of lubricants tested, it is seen that the formulated ester with refrigerant (Est2-F+R134a) provides wear characteristics similar to the mineral oil-R12 mixture.

As with the counterformal contact wear results, R134a alone lacks lubricative properties. Though not shown, very high wear rates, equivalent to testing in air, are observed for tests run with R134a by itself. While the addition of R12 to the mineral oil decreases wear, the addition of R134a to both the PAG and Ester tends to increase wear.

The friction data for the area contact are shown in Fig. 6. In general, these data correlate reasonably well with the wear data. As with the wear, the friction obtained with the formulated esters with R134a compares favorably with the presently used mineral oil-R12 mixture.

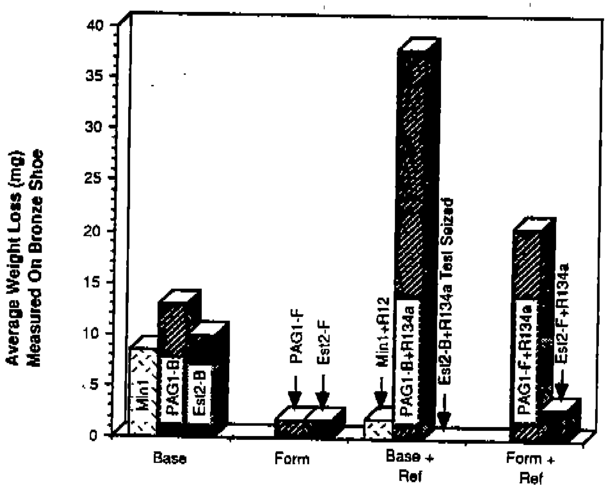

Fig. 5 - Area Contact Wear Results

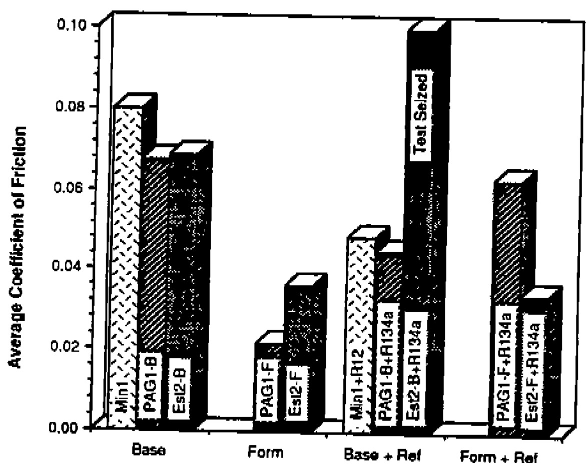

Fig. 6 - Area Contact Coefficient of Friction

Conformal Contact

Wear data for the conformal contact are shown in Fig. 7. As with the area contact results, the mineral oil provides the best wear resistance for this contact. The addition of R12 to the base mineral oil improves wear resistance. XPS analysis is underway to determine if any surface films were formed on either the aluminum pad or the mating steel pin. The addition of R134a to the PAG2-B tends to increase the amount of wear. Although not graphically shown, when R134a was run by itself, seizure occurred after 15 minutes of testing. The difference in the wear results for the two esters is probably due to the fact that generally thinner oil films are generated for low viscosity oils than for high viscosity oils. Viscosity data for both esters are provided in Table 3. For this contact, the PAG oil seems to give the best performance with R134a.

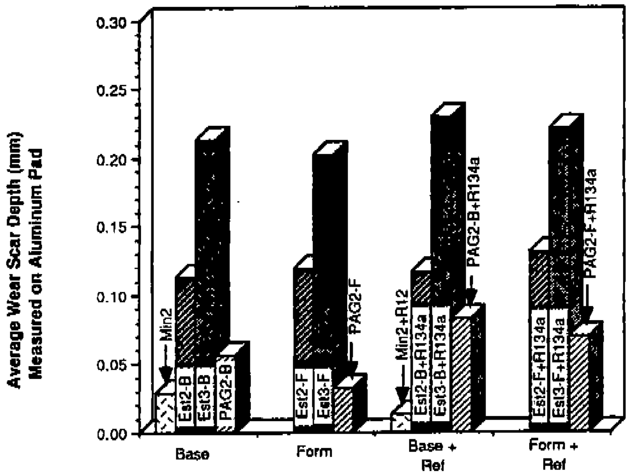

Fig. 7 - Conformal Contact Wear Results

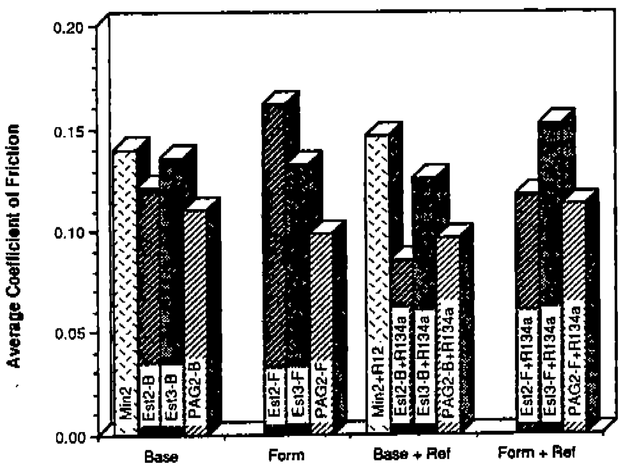

Fig. 8 - Conformal Contact Coefficient of Friction

Fig. 8 shows the coefficient of friction data for the conformal contact. Even though the mineral oil-R12 mixture shows very good wear characteristics compared to the esters and PAG with R134a, its friction characteristics tend to be worse than those of the latter mixtures. Overall, the PAG2-F gives the lowest coefficient of friction.

CONCLUSIONS

The wear resistance of a contact pair depends on many variables. In this paper, some of the more important of these variables have been examined. In general, the formulated oils provide better wear resistance than their base counterparts when tested alone. When refrigerant is added, however, the results are less predictable. R12 in solution with the oil effectively serves to lower the wear observed with the oil alone for all three contact geometries. The production of surface films, especially from the chlorine atom in the R12, serves to decrease wear. R134a, on the other hand, has no chlorine atom and does not exhibit lubricative properties. The relationship between test conditions and reaction rates involving the production of metallic chlorides needs further investigation. Unlike the wear results, the coefficient of friction obtained for oils tested with R134a compares favorably with that obtained for presently used oil-refrigerant mixtures.

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